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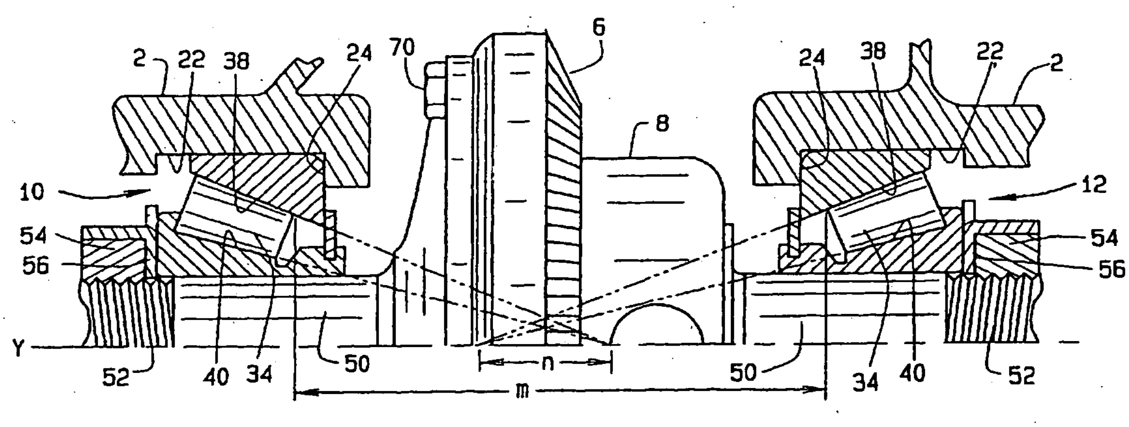
(54) Title: AUTOMOTIVE DIFFERENTIAL



(57) **Abstract:** Each bearing has a cup (30) on one of the seats of the housing (2), a cone on the sourrounding seat (50) of the carrier (8) and tapered rollers organized in a single row between tapered raceways (38, 40) on the cup and cone. The small ends of the tapered rollers are presented toward the interior of the housing. This configuration enables the bearings to retain essentially the same setting, notwithstanding differential thermal expansions. To assemble the differential, the two bearings are installed on the carrier. The distance (a) between the back faces (36) of the cups is measured, as is the distance (b) between shoulders (24) at the ends of the seats in the housing. The carrier is installed in the housing with the cups of its bearings in the bearing seats of the housing. After bringing the ring gear against the pinion, the cone of the one bearing is advanced over its seat on the carrier a distance Δc which imparts the proper mesh to the ring gear and pinions. The cone of the other bearing is advanced over its seat a distance amounting to a-b-Δc+p, where p is the setting desired for the bearings.

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AUTOMOTIVE DIFFERENTIAL

Technical Field

This invention relates in general to differentials for having automotive vehicles and more particularly to a thermally stable differential.

Background Art

The typical differential for an automotive vehicle has (Fig. 1) a housing in which meshed gears rotate, one being a pinion that is connected to the drive shaft for the vehicle and the other a ring gear that is on a differential carrier which rotates in bearings set into the housing. The carrier has a cross shaft on which a pair of beveled gears rotate, and those bevel gears mesh with more bevel gears that are connected to the axle shafts which extend away from the carrier to drive road wheels. The bevel gears connected to the axle shafts have the capacity to rotate within the differential carrier at different angular velocities to compensate for the different angular velocities at which the two axle shafts will rotate when the vehicle negotiates a turn, for example.

Summary of the Invention

Antifriction bearings support the carrier in the housing – one at each end of the carrier – and typically these bearings are single row tapered roller bearings mounted in opposition so that they accommodate thrust loads as well as radial loads and thereby confine the carrier axially. Other bearings having their raceways or raceway contacts oblique to the axis of rotation are suitable as well. Where tapered roller bearings are used, they are normally installed in the direct configuration, that is, with the tapered rollers of the two bearings tapering downwardly away from each other so that the large ends of the rollers for the one bearing are presented toward the large ends of the rollers for the other bearing. Indeed, if the envelopes for the rollers are extended to their respective apices, those apices would lie at two points along the axis of rotation for the bearings – there being one apex point for each bearing –

and the two bearings would be located between those apex points, as would the ring gear. Often the housing is manufactured from a metal having a higher coefficient of expansion than the metal from which the carrier is manufactured. A rise in temperature for such a differential will lead to uneven expansion between the housing in which the cups (outer races) of the bearings are installed and the carrier on which the cones (inner races) of the bearing are installed. This leads to excessive internal clearances in the bearings, which can disrupt the mesh between the pinion and ring gear and produce excessive noise and wear.

<u>Brief Description of Drawings</u>

Figure 1 is a sectional view of a typical automotive differential constructed in accordance with the prior art;

Figure 2 is a sectional view of a differential constructed in accordance with and embodying the present invention;

Figure 3 is fragmentary sectional view of one of the indirectly mounted bearings that support the carrier of the differential;

Figure 4 is a sectional view showing the two bearings around the carrier;

Figure 5 is an elevational view of the carrier separated from the housing;

Figure 6 is an elevational view of the housing absent the carrier;

Figure 7 is a fragmentary sectional view of one of the bearings as it is advanced over its bearings seat; and

Figures 8-10 are sectional views of modified bearings for the differential.

<u>Best Mode for Carrying Out the Invention</u>

Referring now to the drawings, a differential A (Fig. 2) transfers torque from a drive shaft B to two axle shafts C which extend out to road wheels to which they are coupled. The differential A enables the axle shafts C to rotate at different angular velocities while delivering torque to both of them.

In very basic terms, the differential A includes (Fig. 2) a housing 2, a pinion 4, a ring gear 6 driven by the pinion 4 and a carrier 8 to which the ring gear 6 is attached, so that the pinion 4 likewise drives the carrier 8. The pinion 4 rotates about a longitudinal axis X, whereas the ring gear 6 and carrier 8 rotate about a transverse axis Y. The carrier 8 is supported in the housing 2 on two single row tapered roller bearings 10 and 12 which are mounted in opposition – indeed, in the indirect configuration – so as to confine the carrier 8 and ring gear 6 axially along the transverse axis X, while leaving it free to rotate. The housing 2 is cast from a metal, such as aluminum, which has a higher coefficient of thermal expansion than steel from which the carrier 8 is formed. As a consequence, the housing 2 and carrier 8 expand and contract differently with variations in temperature.

The pinion 4 lies at the end of a shaft 18 which rotates in bearings 20 mounted in the housing 2. The bearing 20, while permitting the shaft 18 and its pinion 4 to rotate about the axis X, confine the pinion A axially, so that the pinion 4 assumes a fixed axial position along the axis X.

Along the transverse axis X, the housing 2 contains (Fig. 2) two bearing seats in the form of half bores 22 which open away from the interior of the housing 2. The half bores 22 lead up to shoulders 24 that face away from each other. One half bore 22 contains the bearing 10, while the other contains the bearing 12. Indeed, the bearings 10 and 12 are retained firmly in the half bores 22 by caps 26 which fit over the half bores 22 and are attached firmly to the housing 2 with cap screws 28 (Fig.6).

The bearings 10 and 12 support the carrier 8 in the bores 24 of the housing 2. Each bearing 10 and 12 includes (Fig. 3) an outer race in the form of a cup 30, an inner race in the form of a cone 32, and rolling elements in the form of tapered rollers 34. The cup 30 fits into the half bore 22 for its bearing 10 or 12 within which it is firmly retained by the cap 26 for that half bore 22 and of course the cap screws 28 which hold

the cap 26 in place. When the cup 30 is so disposed, its back face 36 is against the shoulder 26 at the end of the bore 24. The cup 30 also has a tapered raceway 38 that is presented inwardly toward the axis Y. The cone 32 likewise has a tapered raceway 40, but it is presented outwardly away from the axis Y and toward the raceway 38 of the cup 30. At the large end of its raceway 40, the cone 32 has a thrust rib 42 which leads out to a back face 44. The tapered rollers 34 fit between the raceways 38 and 40 of the cup 30 and cone 32, respectively. Their side faces lie along and contact the raceways 38 and 40, there being essentially line contact between the side faces of the rollers 34 and the raceways 38 and 40. The large end faces of the rollers 34 bear against the thrust rib 42, and indeed, the thrust rib 42 prevents the rollers 34 from being expelled from the space between the two raceways 38 and 40. The rollers 34 are on apex, meaning that the conical envelopes in which the side faces of the rollers 34 lie have their apices at a common point along the axis Y (Fig. 4). The envelopes for the two raceways 38 and 40 likewise have their apices at essentially the same point. Beyond the small end of its raceway 40, the cone 32 is fitted with a retainer 46 which projects radially outwardly and along its periphery lies along the back face 36 of the cup 30. Thus, the retainer 46 holds the bearing 10 or 12 together for handling purposes.

The apices for the bearing 10 lie between the small ends of the rollers 34 for the bearing 12 and the apices for the bearing 12 (Fig. 4). The apices for the bearing 12 lie between the small ends of the rollers 34 for the bearing 10 and apices for the bearing 10. In other words, the conical envelopes for rollers 34 and raceways 38 and 40 of the bearing 10 overlap the conical envelopes for the rollers 34 and raceways 38 and 40 for the bearing 12. The small ends of the rollers 34 in the bearing 10 and the small ends of the rollers 34 for the bearing 12 are spaced apart axially a distance <u>m</u>. Preferably the distance <u>n</u> between the apices for

the two bearings 10 and 12 should not exceed 0.5m and better still should not exceed 0.20m.

The carrier 8 for the most part occupies the interior of the housing 2, but has (Fig. 2) cylindrical bearing seats 50 which project into the bearings 10 and 12 – indeed, into the cones 32 of the bearings 10 and 12. Actually, the seats 50 project through the cones 32, and beyond the back faces 44 of the cones 30, they are provided with threads 52 (Fig. 3) with which nuts 54 are engaged, there being between each nut 54 and the back face 44 opposite it a locking washer 56.

Between its two bearing seats 50, the carrier 8 has (Fig. 2) a cross shaft 60, the axis of which is perpendicular to the axis Y. The cross shaft 60 carries bevel gears 62 which mesh with more bevel gears 64, with the latter having journals 66 that project into the carrier 8 where they are free to rotate about the axis Y. The journals 66 are hollow and the axle shafts C extend into them. The journals 66 and the shafts C are engaged at mating splines 68.

The ring gear 6 is attached to the carrier 8 with cap screws 70 (Fig. 2). It meshes with the pinion 4, with the plane in which its pitch circle lies intersecting the axis X at the point between the apices for the two bearings 10 and 12, preferably midway between the two apices. When the pinion 4 rotates on its shaft 18, the carrier 8 revolves about the axis Y. The cross shaft 60 rotates with the carrier 8 and through the meshed bevel gears 62 and 64 rotates the axle shafts C. Normally the two axle shafts C rotate at the same angular velocity, but the arrangement permits one to revolve at a different velocity than the other.

The carrier 8 rotates without any axial free motion and further with the axial position of the ring gear 6 such that the pinion 4 and ring gear 6 properly mesh. To this end, the bearings 10 and 12 operate in a condition of light preload. The nuts 54 control the amount of preload, and also the axial position of the carrier 8 within the housing 2, and that

position determines the axial position of the ring gear 6 and its mesh with the pinion 4.

To assemble the differential A, the carrier 8 is fitted with its cross shaft 60 and bevel gears 62 and 64, and in addition, the ring gear 6 is secured to it with the cap screws 70. Also the bearing 10 and 12 are fitted to the carrier 8, in the indirect configuration, their cones 32 being pressed onto the cylindrical bearing seats 50 (Fig. 5). Since the bearings 10 and 12 are indirectly mounted, the back faces 44 of the cones 32 face away from each other and the small ends of the rollers 34 are presented toward each other as are the back faces of the cups 30. While the cones 32 are pressed over the seats 50, neither is advanced to the position that it finally assumes. The rollers 34 and cups 30 remain around their respective cones 32, inasmuch as the retainers 46 unitize the bearings 10 and 12, at least for purposes of handling. The washers 56 are installed over the threads 52 on the bearing seats 50 and against the back faces 44 of the cones 32. Then the nuts 54 are engaged with the threads 52, and turned down lightly against washers 56.

The housing 2, while containing the pinion 4, otherwise remains vacant (Fig. 6). Moreover, its half bores 22 are exposed, inasmuch as the caps 26 which normally cover them are detached. With the housing 2 and carrier 8 separated, measurements are taken from both. On the carrier 8, one measures the distance a between the back faces 36 on the cups 30 of the two bearings 10 and 12, while exerting axially directed forces on the cups 30 with gage fingers 74 that bear against the back faces 36 of the cups 30 (Fig. 5). The axial force urges the two cups 30 apart and seats the rollers 34 against their raceways 38 and 40 and also against the thrust ribs 42. To insure seating, the carrier 8 may be rotated slowly while the force is applied and the distance a measured. Also the distance b between the shoulders 24 at the ends of the half bores 22 in the housing 2 is measured (Fig. 6).

Once the distances a and b are determined, the carrier 8 and its bearings 10 and 12 are dropped into the housing 2. The ring gear 6 engages the pinion 4 while the cups 30 of the bearings 10 and 12 enter and come to rest in the half bores 22. The caps 26 are installed over the cups 30 and secured with the cap screws 28. As the cap screws 28 are tightened, the cups 30 of the bearings 10 and 12 are urged toward the shoulders 24 at the ends of the half bores 22. The caps 26 retain the cups 30 firmly in the housing 2 with the back faces 26 of the cups 30 against the shoulders 24 at the ends of the half bores 22. The bearings 10 and 12 at this juncture possess considerable end play, and as a consequence, the carrier 8 can shift axially in the housing 2 – indeed, far enough to bring the ring gear 6 snugly against the pinion 4. The carrier 8 is moved axially toward the cup 30 of the bearing 12 until the ring gear 6 bottoms out against the pinion 4, that is, until the carrier 8 can advance no farther toward the cup 30 of the bearing 12. At this juncture, the distance c between a reference surface on the cup 30 for the bearing 10 and a reference surface along the bearing seat 50 for the bearing 10 is measured (Fig.7).

Thereupon, the nut 54 behind the bearing 10 is turned down with a nut driver 76 that fits over the nut 54 and the end of the seat 50 over which it threads (Fig. 7). It moves the cone 32 of the bearing 10 toward the cup 30. In time the rollers 34 seat against the raceways 38 and 40 and the thrust rib 42. Further rotation of the nut 54 causes the ring gear 6 to back away from the pinion 4. The advancement of the nut 56 continues until the ring gear 6 and pinion 4 are properly meshed, and then still further the amount of preload deflection assigned to the bearing 10. When this is achieved, the advancement of the nut 54 ceases and the distance c is again measured. Thereupon, the difference Δc between the two distances c is calculated.

Next the nut 54 behind the bearing 12 is turned down to advance it over its threads 52, indeed, to advance it a distance d. The nut 54, of

course, displaces the cone 32 of the bearing 12 a like amount. One calculates the distance d using the following formula:

$$d = a - b - \Delta c + p$$

The dimension p represents the preload desired for the two bearings 10 and 12, with that preload being expressed in terms of a lineal dimension.

The two nuts 54 are then secured with their locking washers 56, so that they will not rotate on their threads 52, thus fixing the position of the cones 32 for the two bearings 10 and 12 on the carrier 8 and likewise the position of ring gear 6 with respect to the pinion 4.

While the housing 2 and carrier 8 are formed from different metals, with the metal of the former having a greater coefficient of expansion then the latter, the setting for the bearings 10 and 12 remains relatively stable over temperature changes that occur, not only with the seasons, but by virtue of friction generated during operation as well. In this regard, a rise in temperature will cause both the housing 2 and the carrier 8 to expand along the axis Y, thus increasing the spread between the bearings 10 and 12. But the housing 2, being formed from a metal with a higher coefficient of expansion, expands even more along the axis Y, and this tends to increase preload. However, the housing 2 along the axis Y also expands radially and this tends to relax preload. The effects of the two expansions generally offset each other, and the setting of the bearings 10 and 12 remains essentially the same, that is to say, the preload does not undergo a significant change. Indeed, if the apices for the two bearings 10 and 12 coincided, the expansions would exactly offset each other. Since the ring gear 6 and pinion 4 mesh generally between the apices for the rollers 34 of the two bearings 10 and 12, expansion and contraction of the carrier 8 and housing 2 does not disturb, to any significant measure, the mesh between the ring gear 6 and pinion 4.

Modifications are possible. For example, the cones 32, instead of being secured and positioned on the bearing seats 50 with nuts 56, may

be secured by upsetting the end of the seats 50 in the provision of formed ends 80 (Fig. 8) which capture the cones 32 on the seats 50. To provide the adjustment required to position the carrier 8 with the ring gear 6 and pinion 4 properly meshed and with the bearings 10 and 12 set to the proper preload, the integral thrust ribs 42 on the cones 32 are replaced with adjustable ribs 82 fitted to the cones 32 with threads 84. This enables the ribs 82 to move axially over the cones 32 and control the axial position of the tapered rollers 34 along the raceways 38 and 40, and likewise the axial position of the carrier 8 in the housing 2. To secure the adjustable ribs 82, once they are positioned, their threads 84 may be deformed into slots along the threads 84 of the cones 32. The same formula applies.

On the other hand, the adjustment may be effected at the cups 30 of the two bearings 10 and 12. To this end, each cup 30 may be provided with external threads 86 (Fig. 9) leading away from its front face and a nut 88 threaded over the threads 86 and against a surface on the housing 2. The nuts 88 control the seating and position of the ring gear 6. Each nut 88 has a window 90 through which the threads 88 are exposed, and here the threads 88 may be deformed, once the bearings 10 and 12 are set, to prevent the nuts 92 from thereafter turning. Then again, each cup 30 may have external threads 92 (Fig. 10) which engage internal threads 94 in the housing 2. The cup 30 also has external spline 96 over which a lock plate 98 fits, and the lock plate 98 is secured against the housing 2 with a machine screw 100, all to prevent the cup 30 from rotating. By turning the cup 30, one can adjust the axial position of the carrier 8 in the housing 2 and the setting of the bearings 10 and 12 as well. Instead of having external threads 92 on the cups 30, they may be on adjustment sleeves 102 (Fig. 11), which engage internal threads 104 in the housing 2 and bear against the back faces 36 of the cups 30 in lieu of the fixed shoulders 24. By rotating the sleeves 102, one can control the axial position of the carrier 8 and the setting of the

bearings 10 and 12. The housing 2 contains lock plates 106 secured by machine screws 108 to fix the position of the sleeves 102 once the proper position for the carrier 8 and setting for the bearings 10 and 12 is achieved. Again, the same formula applies.

In lieu of the carrier 8 being supported on the tapered roller bearings 10 and 12, the carrier 8 may be supported on other types of antifriction bearings which have the capacity to transfer axial as well as radial loads, angular contact ball bearings, for example.

Claims:

1. A differential comprising: a housing; a pinion mounted in the housing for rotation relative to the housing about a first axis; a carrier mounted in the housing for rotation relative to the housing about a second axis, the carrier being formed from a material having a coefficient of thermal expansion that is less than the coefficient of thermal expansion for the material from which the housing is formed; a ring gear on the carrier and meshing with the pinion; and first and second antifriction bearings supporting the carrier in the housing for rotation in the housing about the second axis, each bearing having inner and outer raceways that are oblique to the second axis and are inclined downwardly toward the ring gear and rolling elements arranged in a row between the inner and outer raceways, the inner raceways being on the carrier and the outer raceways being in the housing, the bearings being adjustably mounted to set the mesh between the pinion and ring gear and to set preload into the bearings.

2. A differential according to claim 1 wherein the inner raceways are adjustably mounted on the carrier.

3. A differential according to claim 1 wherein the outer raceways are adjustably mounted in the housing.

4. A differential according to claim 1 wherein the raceways are conical and the apex locations of the raceways are roughly equidistant from the axial location of the mesh between the pinion and ring gear.

5. A differential according to claim 1 wherein an adjustable rib ring allows the bearing location adjustment on the carrier.

6. A differential according to claim 1 wherein a retainer maintains the bearing inner and outer races together during handling.

7. A differential according to claim 1 wherein the inner raceways are on inner races that fit over the carrier and the outer

raceways are on outer races that fit into the housing, and the inner or outer raceways are adjustable axially along the axis.

8. A differential according to claim 7 wherein the inner races are adjustable on the carrier along the second axis.

9. A differential according to claim 7 wherein the outer races are adjustable in the housing along the second axis.

10. A differential according to claim 7 wherein the raceways lie in conical envelopes and the rolling elements are tapered rollers having side faces that lie in conical envelopes.

11. A differential according to claim 10 wherein the conical envelopes for the tapered rollers of the first bearing have their apices essentially at a common point along the second axis, and the conical envelopes for the tapered rollers of the second bearing have their apices essentially at a common point along the second axis.

12. A differential according to claim 11 wherein the apices for the rollers of the first and second bearings are located between the small ends of the rollers for the first and second bearings.

13. A differential according to claim 12 wherein the small ends of the rollers for the first and second bearings a spaced axially apart a distance <u>m</u>; wherein the apices for the rollers of the first and second bearings are spaced axially apart a distance <u>n</u>; and wherein <u>n</u> does not exceed about 0.6 m.

14. A differential according to claim 13 wherein <u>n</u> does not exceed about 0.2m.

15. A differential according to claim 12 wherein the pitch circle for the ring gear lies in a plane that intersects the second axis generally midway between the apices for the rollers of the first and second bearings.

16. A differential comprising: a housing; a pinion mounted in the housing for rotation relative to the housing about a first axis; a carrier mounted in the housing for rotation relative to the housing about a

second axis, a ring gear on the carrier and meshing with the pinion; and first and second antifriction bearings supporting the carrier in the housing for rotation in the housing about the second axis, each bearing having inner and outer raceways that are conical and are inclined downwardly toward the ring gear, and tapered rollers arranged in a row between the inner and outer raceways, the inner raceways being on the carrier and the outer raceways being in the housing, the tapered rollers of each bearing defining conical envelopes having their apices at essentially a common point along the second axis, the apices for the first and second bearings lying between the rollers for the first and second bearings, the bearings being adjustably mounted to set the mesh between the pinion and ring gear and to set preload into the bearings.

17. A differential according to claim 16 wherein the small ends of the rollers for the first and second bearings are spaced axially apart a distance <u>m</u>; wherein the apices for the rollers of the first and second bearings are spaced axially apart a distance <u>n</u>; and wherein <u>n</u> does not exceed about 0.5m.

18. A differential according to claim 17 wherein <u>n</u> does not exceed about 0.2m.

19. A process for assembling a differential having a housing provided with bearing seats, a carrier in the housing and having bearing seats generally opposite the bearing seats in the housing, and first and second bearings defining an axis, with each bearing having an inner race located around one of the bearing seats on the carrier and provided with an inner raceway presented away from the axis, an outer race in one of the bearing seats of the housing and having another raceway presented inwardly toward the axis and the inner raceway, the raceways of the first bearing being oblique to the axis and inclined downwardly toward the second bearing and the raceways of the second bearing being oblique to the axis and inclined downwardly toward the first bearing, said process comprising: fitting the first and second bearings to

the carrier with their inner races over the bearing seats on the carrier but spread farther apart than suitable for operation of the differential, measuring the axial distance (a) between two corresponding races of the first and second bearings, measuring the axial distance (b) between surfaces along the bearing seats against which the corresponding races are fitted; moving the carrier to a position in which the ring gear is against the pinion; moving the other race of the first bearing along its bearing seat to back the ring gear away from the pinion to a position in which the ring gear and pinion properly mesh; measuring the distance (Δc) that the other race of the first bearing moves over its seat; moving the other race of the second bearing axially over its seat a distance amounting to

$$a - b - \Delta c + p$$

where p is the setting, in terms of a lineal dimension desired for the bearings.

20. The process according to claim 19 wherein the bearing seats on the housing lend away from shoulders; wherein the outer races of the bearings have back faces which fit against the shoulders when the carrier is in the housing; wherein the dimension (a) is the distance between the back faces of the outer races before the carrier is installed in the housing; wherein the dimension (b) is the distance between the shoulders at the seats in the housing; and wherein the dimension (Δc) is the distance the inner race is advanced over its seat on the carrier.

21. The process according to claim 20 wherein the inner race of the first bearing is advanced over its seat with a nut threaded onto the carrier.

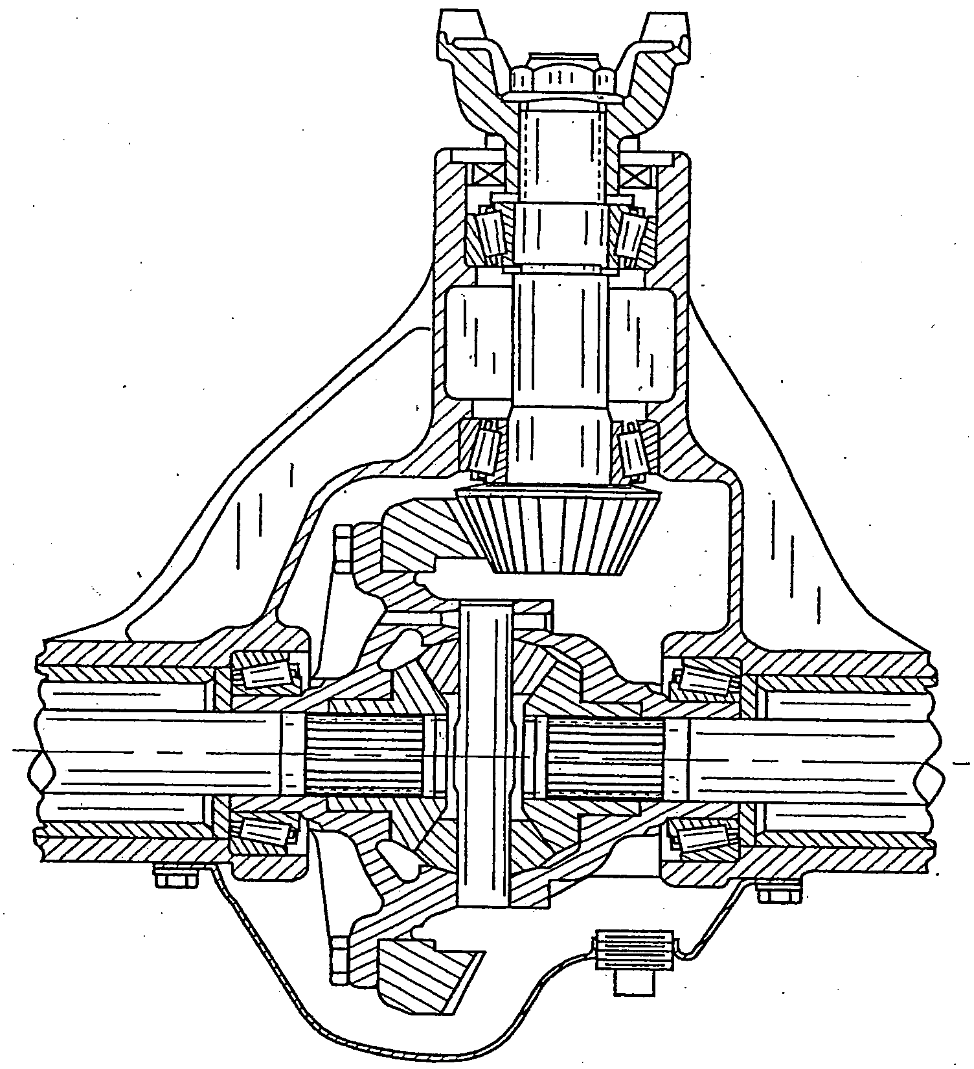

FIG. 1
PRIOR ART

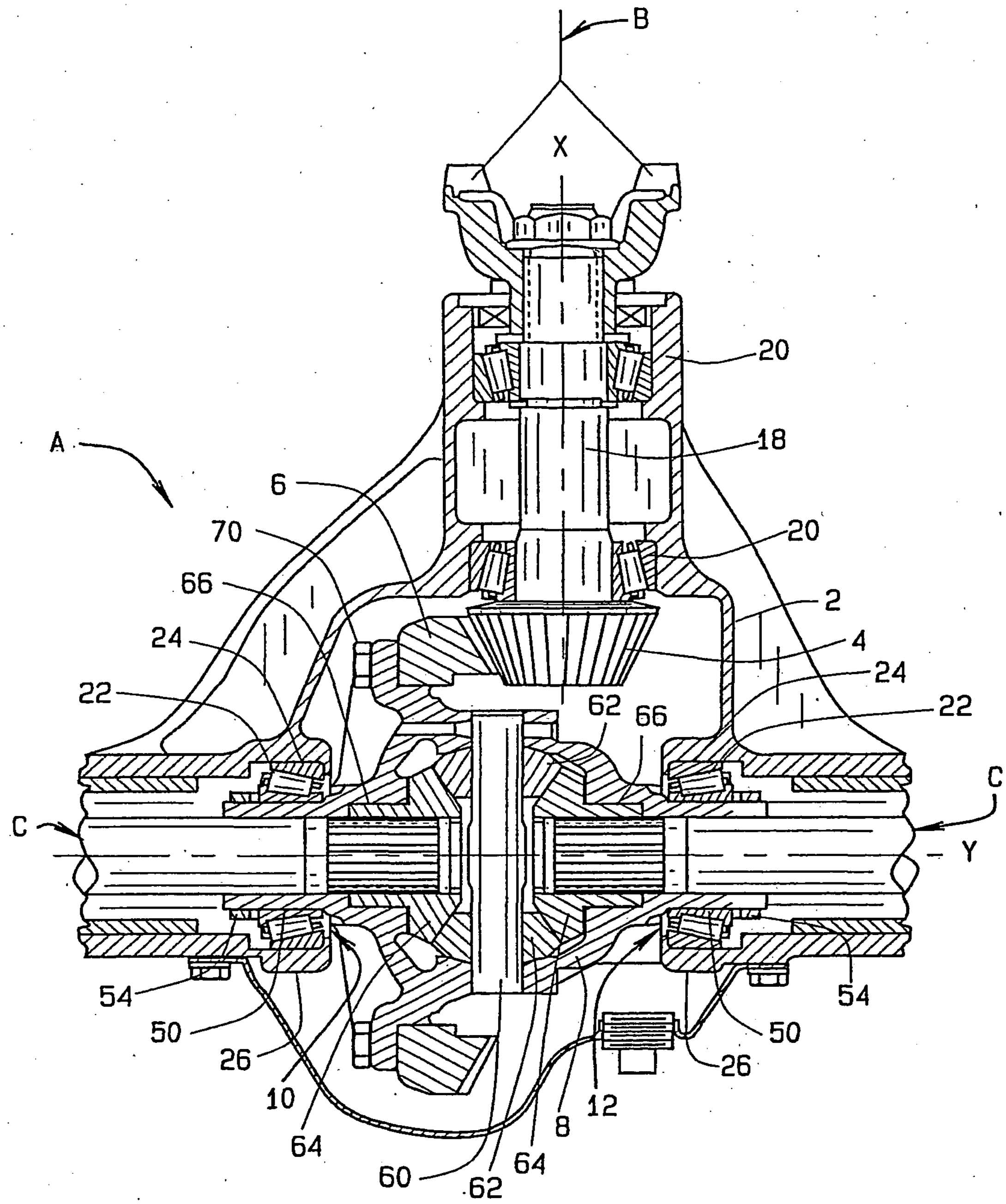
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FIG. 2

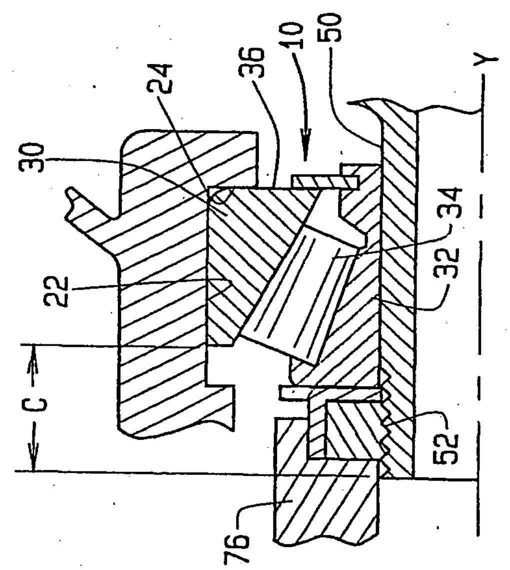

FIG. 7

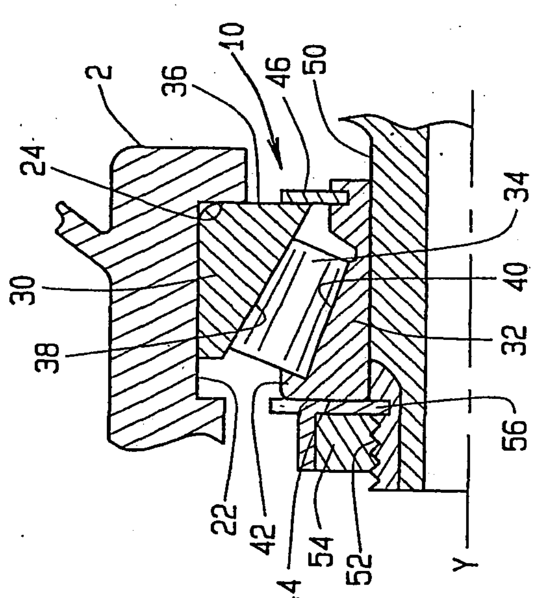

FIG. 3

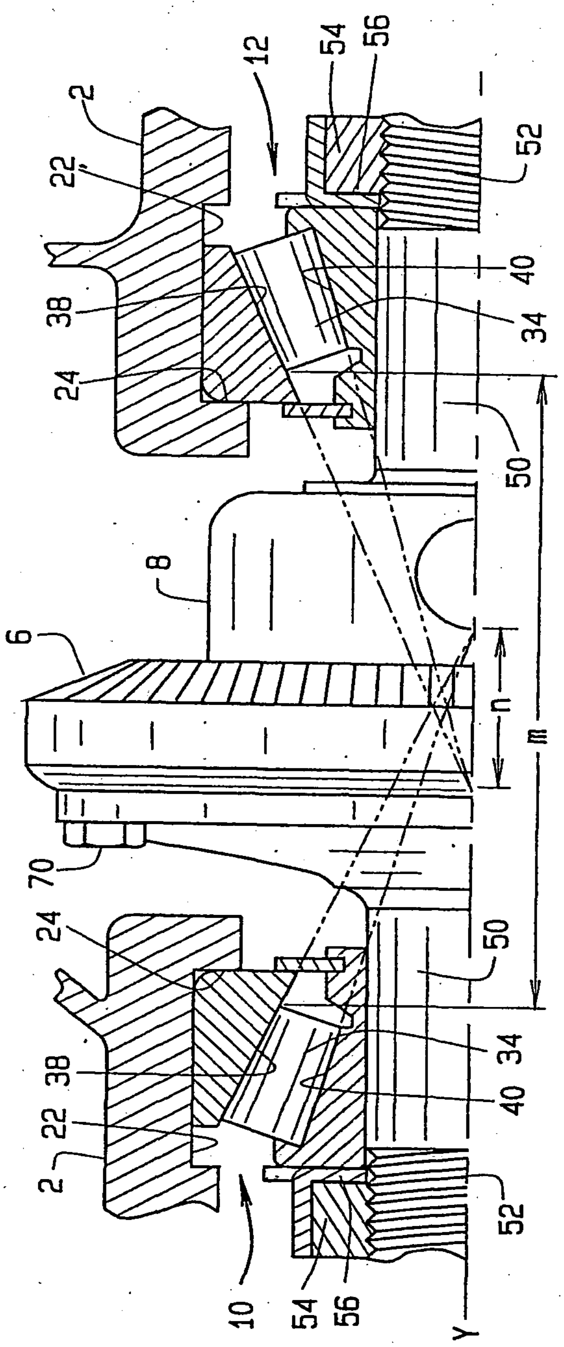

FIG. 4

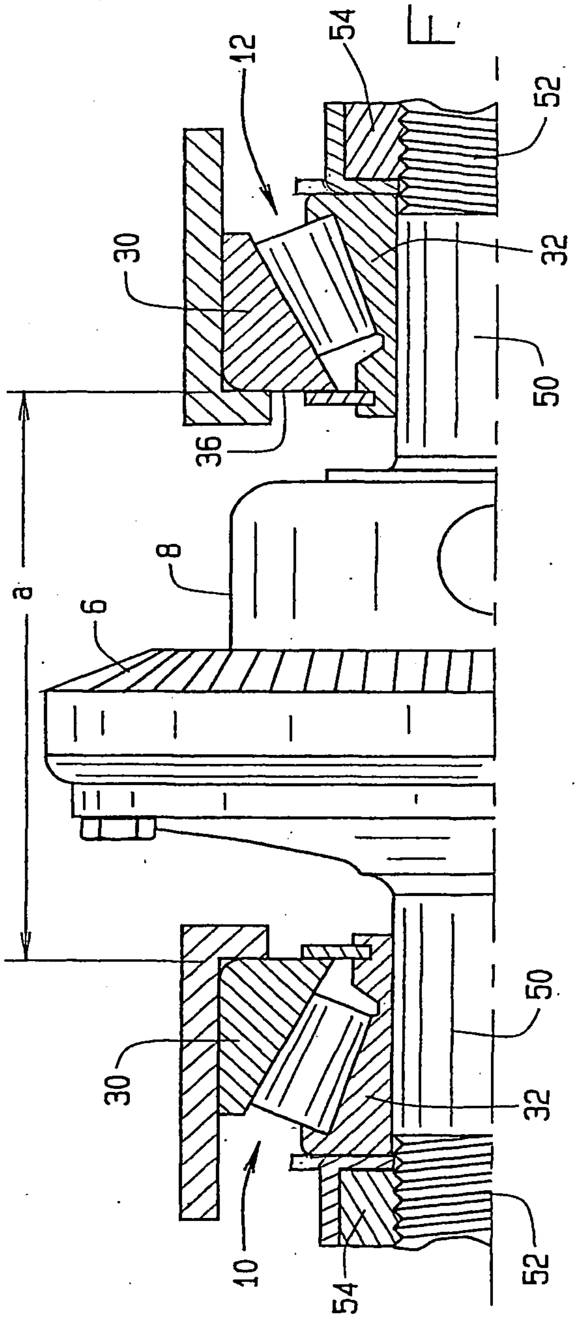

FIG. 5

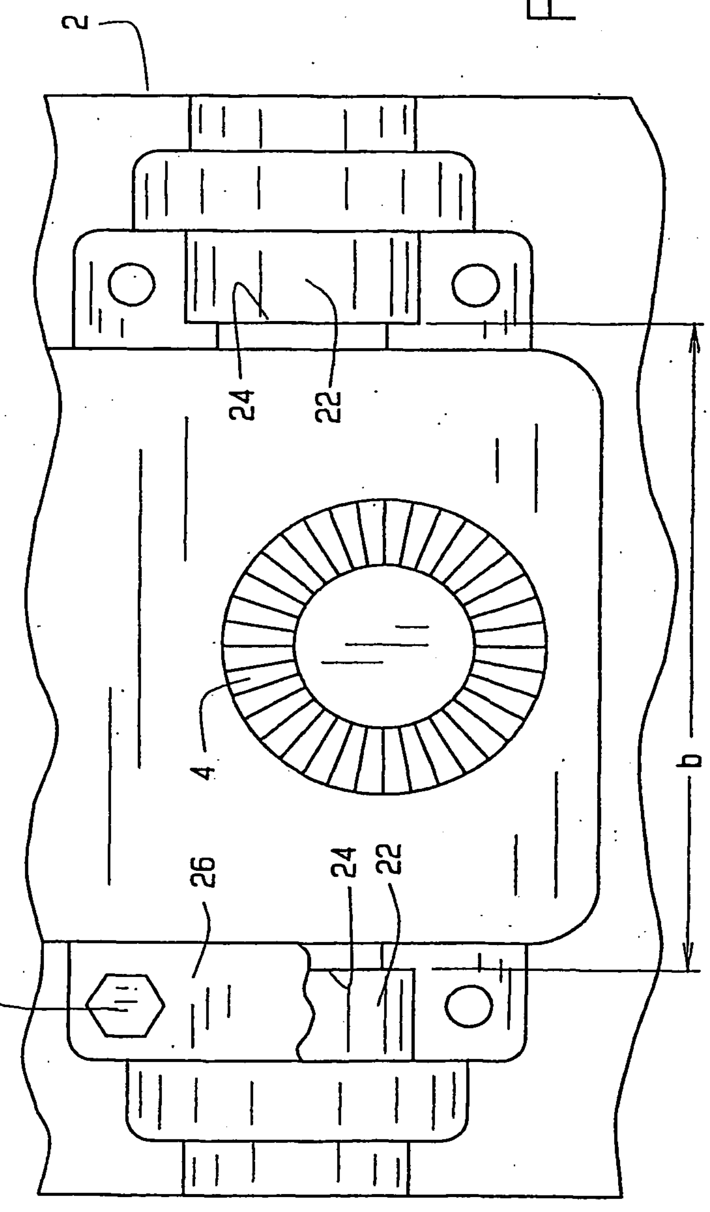

FIG. 6

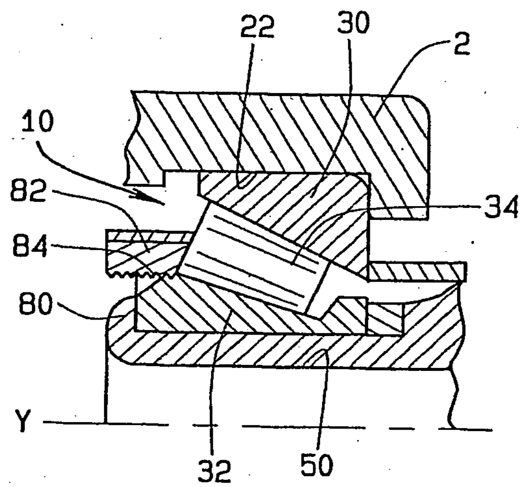

FIG. 8

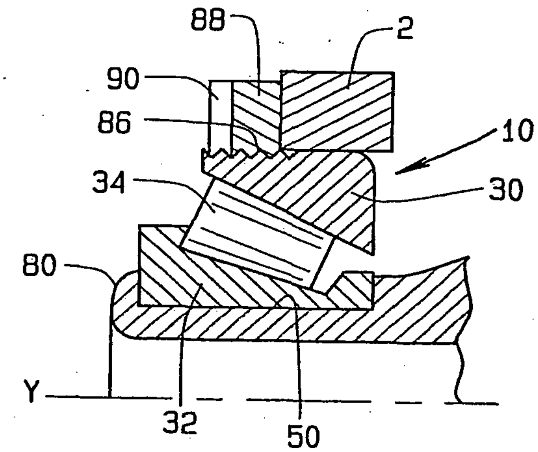

FIG. 9

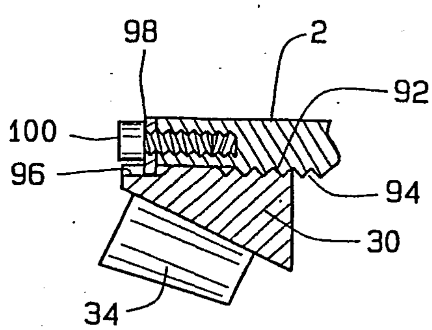

FIG. 10

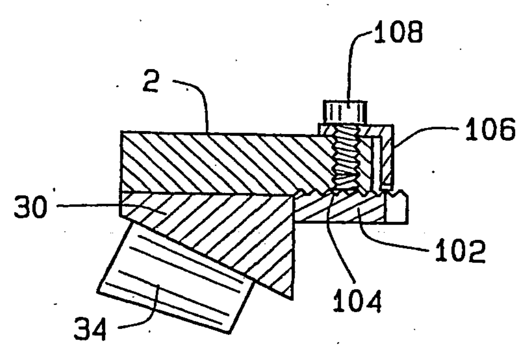

FIG. 11

INTERNATIONAL SEARCH REPORT

Ir   ational Application No
PCT/US 02/15063

A. CLASSIFICATION OF SUBJECT MATTER
IPC 7 F16H48/08

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 7 F16H

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

PAJ, EPO-Internal, WPI Data

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X Y	US 3 202 466 A (EUGENE KAPTUR ROBERT) 24 August 1965 (1965-08-24) the whole document	1,2,5-11 3,4, 12-18
Y	US 3 001 842 A (BOYD WILLIAM E) 26 September 1961 (1961-09-26) figures	3
Y A	US 2 588 670 A (TERRY SYDNEY L) 11 March 1952 (1952-03-11) figures	4,12-18 19
A	EP 0 347 247 A (TIMKEN CO) 20 December 1989 (1989-12-20) figures	11

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[X] Further documents are listed in the continuation of box C.

[X] Patent family members are listed in annex.

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Date of the actual completion of the international search	Date of mailing of the international search report
19 August 2002	26/08/2002

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL – 2280 HV Rijswijk Tel. (+31-70) 340-2040, Tx. 31 651 epo nl, Fax: (+31-70) 340-3016	Goeman, F

C.(Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

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A	US 3 703 769 A (PFEIFFER RICHARD D) 28 November 1972 (1972-11-28) figure 34	19
A	US 4 793 211 A (SCHMIDT JOHANNES) 27 December 1988 (1988-12-27)	
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INTERNATIONAL SEARCH REPORT

Information on patent family members

In tional Application No
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